Proceedings of ASME Turbo Expo 2011
GT2011
June 6-10, 2011, Vancouver, British Columbia, Canada

GT2011-46627

INFLUENCE OF THE DIFFERENT DESIGN PARAMETERS TO THE CENTRIFUGAL COMPRESSOR TIP CLEARANCE LOSS

Teemu Turunen-Saaresti*
Laboratory of Fluid Dynamics
Institute of Energy Technology
Lappeenranta University of Technology
Lappeenranta, Finland
Email: teemu.turunen-saaresti@lut.fi

Ahti Jaatinen
Laboratory of Fluid Dynamics
Institute of Energy Technology
Lappeenranta University of Technology
Lappeenranta, Finland
Email: ahti.jaatinen@lut.fi

ABSTRACT

In this paper the effect of the tip clearance was studied with six different centrifugal compressors and data available in literature. The changes in the overall performance of the compressor stage were examined. The aim was to study the influence of the different design parameters to the tip clearance loss. It was evident by the previous studies that the sensitivity of the centrifugal compressor to the tip clearance loss varies with different designs. However, for the designer it is important to know the effect of the tip clearance loss in order to initially evaluate the quality of different designs. Analysis of the data demonstrated that no clear correlation between the sensitivity of the tip clearance loss and the specific speed, the diffusion ratio, the blade number and the ratio of blade heights exists.

NOMENCLATURE

Latin alphabet

Re	Reynolds number	–
a	Coefficient	–
b	Blade height	m
CR	Relative tip clearance ratio	–
d	Diameter	m
DR	Diffusion ratio	–
N	Number of blades	–
N_s	Specific speed	–
P_{bp}	Back plate loss per unit mass	J/kg
q_m	Mass flow through the rear disk clearance	kg/s
r	Radius	m
t	Tip clearance	m
U	Peripheral velocity	m/s

Greek alphabet

η	Efficiency	–
ρ	Density	kg/m^3

Subscripts

0	Zero tip clearance	–
1	Rotor inlet	–
2	Rotor outlet	–
u_2	Peripheral velocity at rotor outlet	–

INTRODUCTION

The influence of the tip clearance on the centrifugal compressor performance has received extensive attention in the literature. Numerous articles discuss mainly axial change of the tip clearance at different mass flows, pressure ratios and rotational speeds. Several correlations taking account the effect of tip clearance to the centrifugal compressor performance have been presented.

In the paper [1] the tip clearance of centrifugal compressors were studied. The clearance was altered by shimming the shroud wall i.e. the axial clearance was changed. The effect of the tip clearance was examined using six different compressors

*Corresponding author

and the clearance ratio (the clearance divided by the rotor exit width t/b_2) was changed from 0.01 to 0.1. The efficiency drop was plotted versus the clearance ratio. The slope of the efficiency drop curve was different with the different cases. The slope of the efficiency drop curve varied from 0.03 to 0.1 efficiency drop per 1 percent change in the clearance ratio. Unfortunately, no detailed data about tested compressors were found.

The effect of the tip clearance on the performance of the centrifugal compressor were studied experimentally and numerically in [2]. The tip clearance was changed by moving compressor casing axially. The clearance ratio was changed from 0.012 to 0.11. The slope of the efficiency drop changed as the operating point of the compressor was changed. The larger the mass flow, the higher the slope of the efficiency drop curve i.e. the efficiency drop due to the tip clearance was larger at the higher mass flow. The effect of the Reynolds number to the efficiency drop due to the tip clearance was found to be minor. The correlation to the tip clearance loss was derived and it was a function of the relative clearance (the average clearance divided by the hydraulic mean diameter), Reynolds numbers Re_{u_2} and Re_t, the peripheral speed, the leakage flow rate, the weighted flow rate within an endwall boundary layer and the weighted flow rate within an impeller flow channel. The change of the efficiency due to the tip clearance height was found to be different under the different compressor operating conditions.

The centrifugal compressor with the pressure ratio of 6 was studied in [3]. The axial tip clearance of the compressor was varied from 3.9 to 11.3 percent of the impeller exit height. The adiabatic efficiency and the total-to-total pressure ratio of the compressor stage and the impeller were plotted with different tip clearances. The performance of the compressor stage was studied using the vaned diffuser and the performance of the impeller was studied using the vaneless and the vaned diffuser. The change in the performance due to the tip clearance was monitored with different rotational speeds and mass flows. Increased tip clearance decreased the efficiency and the pressure ratio of the compressor stage and the impeller at all flow rates. The impeller work was changed only slightly by the clearance change. The efficiency of the whole compressor was almost same with the two highest tip clearances, 8.9 and 11.3 percent. The actual/running tip clearance was also measured with different rotational speeds, and the higher the rotational speed, the smaller the tip clearance.

The axial tip clearance of a centrifugal compressor was varied from 7.6 to 20.7 percent of the impeller exit height in [4]. Similar experiments as mentioned in [3] were conducted. The compressor stage adiabatic efficiency increased when the tip clearance was decreased at all rotational speeds. Also the pressure ratio increased when the tip clearance decreased, but there was hardly any change below 70% rotational speeds. The impeller peak efficiency increased from 0.767 to 0.813 as the tip clearance decreased from 20.7 percent to 8.4. Also the pressure ratio of the impeller increased. The work factor was only slightly increased with decreasing the tip clearance.

The tip clearance of a centrifugal compressor of an APU engine was investigated by altering the axial clearance in [5]. Seven different relative tip clearance ratios, $CR = t/(t+b)$, varying from 0.0072 to 0.127, were tested. The total-to-total pressure ratio was plotted at the different mass flows and the different tip clearance ratios. The decreased tip clearance increased the total-to-total pressure ratio at all mass flows. The isentropic efficiency of the compressor stage was plotted at the different rotational speeds. The isentropic efficiency was increased with the decreased tip clearance at all rotational speeds.

The two-stage centrifugal compressor with the pressure ratio of 14 and the design mass flow of 3.3 kg/s was designed in [6]. Among other things, the tip clearance effect on the performance of the second stage was studied. Two different impellers were tested, one with 16 main and 16 splitter blades with a nearly uniform meridional blade loading and another with 19 main and 19 splitter blades with reduced loading at the impeller exit. It was concluded that the reduced loading at the impeller exit influenced the tip clearance sensitivity of the compressor stage i.e. the slope of the efficiency curve versus the tip clearance was smaller with the impeller with the reduced load.

The well known relationship [7,8] presented for the tip clearance loss is the empirical formula

$$-\frac{\Delta\eta}{\eta_0} = \frac{2at}{b_1+b_2} \tag{1}$$

This formula gives linear variation of the efficiency drop due to the tip clearance which is dependent only of the amount of the tip clearance, the blade heights and the coefficient a. However, if one examines the efficiency drops reported in the literature (see fig. 2), the slope of the efficiency drop varies and the constant value for the coefficient a can not be determined. This means that the efficiency and the pressure changes due to the tip clearance can not be described only using the amount of the tip clearance. This kind of conclusion was also made in [9], and one generated more sophisticated loss correlations. These loss correlations were taking into account the pressure losses due to the leakage, the pressure gradient on the annular clearance and the pressure gradient on blockage in the channel. The loss correlations were adjusted using values of the contraction coefficient of the leakage flow and the slip coefficient. The correlations were compared to measured data (e.g. [3] and [4]) and reasonable agreement was seen. In [9] it was concluded that the slope of the efficiency versus the tip clearance ratio is smaller when the flow rate is reduced, and compressors with a higher design pressure ratio are less sensitive for the tip clearance change i.e. the slope of the efficiency curve is smaller.

The flow field in the centrifugal compressor is complex and it is usually described using the primary and the secondary flow or so-called jet-wake flow pattern. This was introduced by [10]. In order to describe this kind of flow field pattern, the two-zone model is used for the preliminary design of the centrifugal compressors. The primary zone is assumed to be isentropic and losses take place in the secondary zone. The primary zone is located close to the hub near the pressure side of the blade and the secondary zone close to the shroud near the suction side of the blade. This kind of flow pattern has been described also in many experiments ([11–13]), and it has been noticed that the area of the primary and the secondary flow are not universally same in all centrifugal compressors.

The forces influencing to the primary and the secondary flow pattern in the centrifugal compressor are numerous. These can be e.g. the impeller geometry (blade trailing edge angle, passage geometry, etc.), the tip clearance and the diffuser geometry. It is experimentally rather hard to identify the effect of different parameters on the centrifugal compressor flow pattern and loss generation because numerous different kind of geometries and measurements should be made and changes can be within measurement accuracy. Therefore, the most of the experiments presented in the literature show the overall performance change of the compressor stage due to the tip clearance. Some of these experiments are reviewed above.

Detailed flow field in a centrifugal compressor vaneless diffuser inlet has been measured in [14]. Two different tip clearances (clearance ratios t/b_2 of 4.5% and 12.7%) was used. The tip clearance was adjusted by moving the shroud wall axially and the diffuser height was also changed. The pressure ratio of the compressor improved with the reduced tip clearance, and the stall inception was moved to lower mass flows. The LDA measurements were made using one mass flow rate, and the comparison of the measurements with the different tip clearances was made at the radius ratio of $r/r_2 = 1.05$. The flow pattern was found to be rather different with the different tip clearance ratios. With the reduced clearance there was no flow reversal seen at the shroud, and the tangential velocity at the wake was smaller. Also the wake was located at the middle of the passage (in the axial direction) with the reduced clearance, whereas the wake covered larger area and it was pushed more towards the hub because of the tip clearance flow with the larger clearance. In both tip clearance configurations the tip clearance flow was more pronounced in the vicinity of the splitter blade.

In this study the effect of the axial tip clearance with six different centrifugal compressor were studied. The change of the overall performance of the compressors was analysed together with the data available in the literature. The sensitivity of the compressor performance to the tip clearance was found to variate in different compressors, and the change was found to be linear.

MEASUREMENT ARRANGEMENTS

The effect of the tip clearance for six different impellers were conducted by measuring the isentropic efficiency and the pressure ratio of the compressor stage. The compressor stage consists of the inlet cone, the impeller, the vaned or the vaneless diffuser, the volute and the exit cone. The efficiency and the pressure ratio measurements are made according to the test standard [15].

The tip clearance of the compressor was changed axially using the active magnetic bearings which enable accurate control over the axial movement. The accuracy of the magnetic bearing control was 0.01 mm. This way the axial tip clearance could be changed without altering the diffuser geometry. The layout of the experimental set-up is shown in Fig. 1.

However, as the tip clearance was decreased the clearance between the back plate and the casing was increased and vice versa. This has an effect on the performance of the compressor stage. According to [16] and [17] the loss between rotating disk is hardly affected by clearance between them. However, the loss is affected by the mass flow through the slit as seen from correlation presented in [16]

$$P_{bp} = \frac{0.0402}{\mathrm{Re}_{u_2}^{1/5}} \rho_2 \left(\frac{U_2}{r_2}\right)^3 \frac{r_2^5}{4q_m} \tag{2}$$

Therefore, moving the impeller towards the shroud will decrease losses due to the rear disk because of the larger clearance and the larger mass flow through it. In the results this should be seen as the larger efficiency increase than without the change in the rear disk clearance for cases 1, 4 and 6 and both larger and smaller efficiency change in cases 2, 3 and 5. However, there was not seen any difference in the slope of the efficiency curve in cases 2, 3 and 5 when monitoring the efficiency change while moving the impeller towards the shroud or towards the hub.

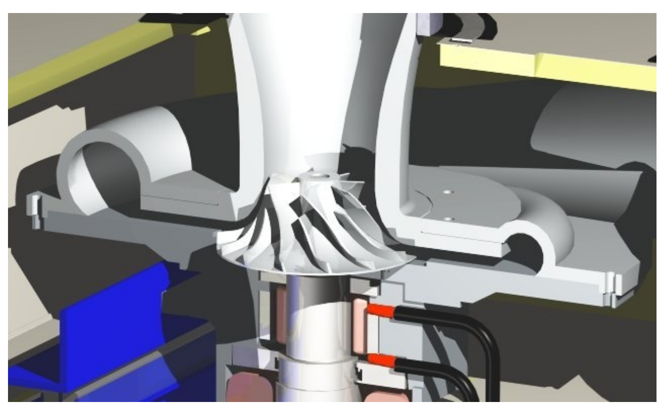

Figure 1. Layout of experimental set-up.

The main parameters of the measured impellers are shown in table 1. All impellers were designed with the backsweep and to have splitter blades. The backsweep angle was same in each impeller. The tip clearance was changed either decreasing it or/and increasing it. For the cases 1,4 and 6 the tip clearance was only decreased from the nominal value, and for the cases 2, 3 and 5 the tip clearance was both increased and decreased from the nominal value. The range of the measured tip clearance ratios are shown in table 2.

Table 1. Parameters of measured compressor configurations

Case	d_2 [mm]	b_2 [mm]	t_2/b_2 [-]	DR [-]	N_s [-]	N	diffuser type
1	162	6.4	0.109	1.40	0.83	18	vaneless
2	120	4.3	0.140	1.40	0.63	18	vaneless
3	158	9.1	0.055	1.82	0.85	20	vaned
4	128	5.7	0.123	1.62	0.59	22	vaned
5	158	8.7	0.057	1.82	0.85	20	vaned
6	128	5.9	0.102	1.62	0.59	22	vaned

Table 2. Measured range of tip clearances

Case	measured range of t_2/b_2
1	0.066 - 0.105
2	0.113 - 0.165
3	0.042 - 0.062
4	0.095 - 0.123
5	0.049 - 0.062
6	0.081 - 0.099

The tip clearance ratios shown in tables 1 and 2 are calculated using the measured cold clearances. However, the clearances in the compressor tends to change when the compressor is operated. The tip clearances under operation were monitored in cases 3 and 4. It was noticed that the running clearance was smaller than the cold clearance and the higher peripheral speed decreased the running clearance more than the lower peripheral speed. The results in this paper are presented using the cold clearances, as they also are used in other papers in literature.

CHANGE OF EFFICIENCY VERSUS TIP CLEARANCE

The data gathered from the literature ([1–6]) and the measurements conducted in this study are presented in Fig. 2. The efficiency plotted in the figure is for the compressor stage. However, the different efficiency is reported in different references. In [3] and [4] the efficiency is calculated dividing the total ideal enthalpy rise with the actual total enthalpy rise, in [2] the adiabatic efficiency is shown, in [5] the isentropic efficiency is utilised and there is no indication what efficiency is used in [1] and [6]. The outlines of the compressors used in the literature are shown in table 3.

In [3] and [4] the compressor stage efficiency is plotted using four different tip clearance ratios and six different rotational speeds. The efficiencies were found to be same in the two largest tip clearances in [3]. The efficiencies are collected at each rotational speed at the same mass flow and the mass flow is chosen at the peak efficiency. The variation of the efficiency versus the tip clearance is changing rather linearly. However, the slope of these efficiency curves at the different rotational speeds is different. The higher the velocity, the larger the slope i.e. the change of the efficiency versus the tip clearance is larger with higher rotational speeds. In order to plot the results from the different references in the same figure, the efficiency of the zero clearance is estimated approximating linear change of the efficiency versus the tip clearance.

There is a large variation in the slopes of the efficiency. It varies from 0.02 to 0.10 efficiency drop per 0.1 change in the relative tip clearance. Also the curves from the different sources are rather linear. There is a slight nonlinearity in some data at small relative tip clearances. The equation 1 behaves linearly and the coefficient in equation is chosen to be 0.9 by [7], and [8] suggested coefficient to be from 1.5 to 3.0. All these values will be within the shown range of the efficiency slope. If the efficiency in equation 1 is chosen to be $\eta_0 = 0.8$ and the ratio of blade heights is chosen to be $b_1/b_2 = 7.5$, the coefficient should be in range $a = 0.9...5.1$ in order to be within the range shown in Fig. 2. The deeper analysis of the results is needed in order to evaluate the value of the coefficient a.

SLOPE OF EFFICIENCY

From the curves shown in Fig. 2 the slope of each curve can be determined. It is done by assuming the linear behaviour of the curves. When a centrifugal compressor stage is to be designed, one needs to choose certain parameters. Depending on design methodology used or on designers customs different parameters are used. These might be e.g. the diffusion ratio and the specific speed. In order to determine whether there is any dependence in the slope of the efficiency curves versus the tip clearance, the values of the slopes are plotted as a function of the specific speed, the diffusion ratio, the number of blades and the ratio of rotor inlet and outlet blade heights.

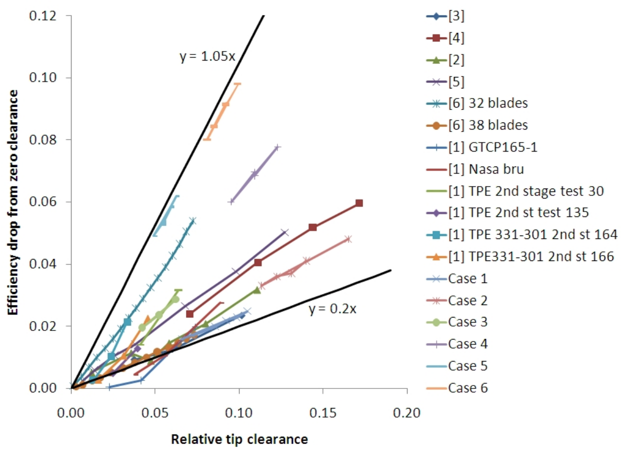

Figure 2. Change of efficiency of compressor stage with different relative tip clearances.

Table 3. Outlines of compressor data from literature

Ref	d_2 [mm]	b_2 [mm]	t_2/b_2 [-]	DR [-]	N_s [-]	N	diffuser type
[3]	161	5.2	0.039	1.39	0.72	19	vaned
[4]	137	4.7	0.027	1.22	0.77	24	vaned
[2]	154	8.0	0.050	1.39	0.44	14	-
[5]	-	-	0.060	-	-	-	vaned
[6]	-	-	-	-	-	32 / 38	vaned
[1]	-	-	-	-	-	-	-

The slope of efficiency versus the specific speed is shown in Fig. 3. The data available in the literature and measurements made in this study are shown. There is little or no correlation of the slope against the specific speed. The slope seems to be little bit larger with the larger specific speed. However, the correlation is so weak that the specific speed can not be used as a factor to determine the slope of the efficiency. In Fig. 4 the slope of the efficiency is plotted against the diffusion ratio. The diffusion ratio is a ratio of relative velocities at the compressor inlet tip and the compressor outlet. There is little or no correlation between the slope of the efficiency and the diffusion ratio. The compressors with the higher diffusion ratio seems to be more sensitive for the tip clearance effects. However, there is still rather high deviation in the slope. The slope of the efficiency against the rotor blade number and the blade height ratio, b_1/b_2, is shown in Fig. 5 and 6. Also the dependency of the efficiency from these factors is not unambiguous.

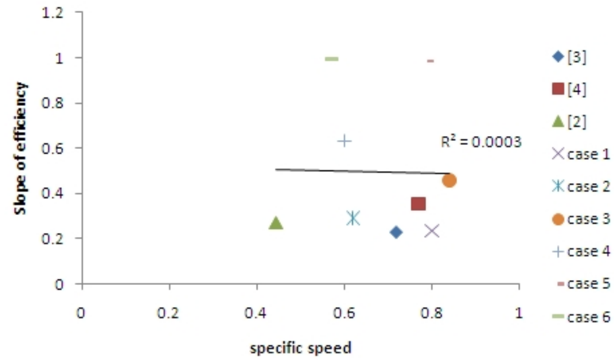

Figure 3. Slope of efficiency curves versus specific speed.

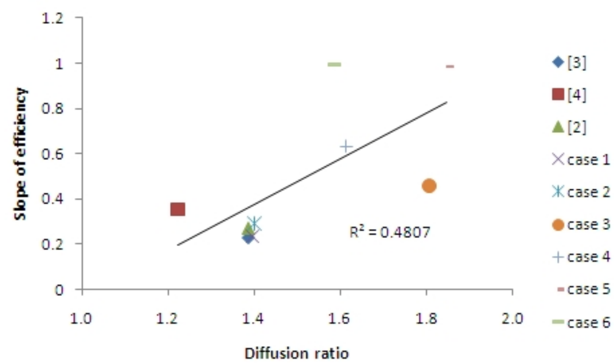

Figure 4. Slope of efficiency curves versus diffusion ratio.

CHANGE OF PRESSURE RATIO VERSUS TIP CLEARANCE

The change of the pressure ratio versus the tip clearance is shown in Fig. 7. The data from [3–5] and from measurements conducted in this study are shown. In other references data about the pressure ratio drop due to the tip clearance is not found. The pressure ratio is also affected by the tip clearance. The larger the tip clearance the higher is the reduction in the pressure ratio. As seen in the efficiency, also the pressure ratio relation to the tip clearance is rather linear in all cases, and the difference in the slope of the pressure ratio curve is significant. In order to examine the effect of the tip clearance to the pressure ratio in more

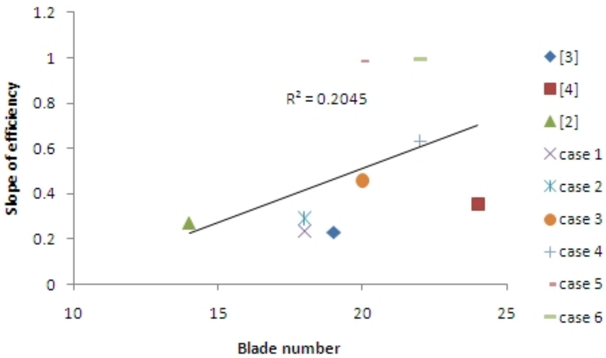

Figure 5. Slope of efficiency curves versus number of blades.

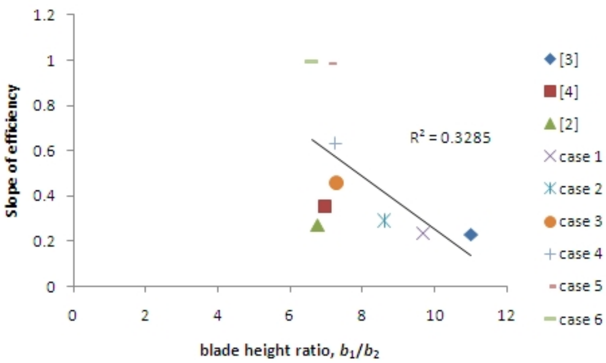

Figure 6. Slope of efficiency curves versus ration blade heights.

detail, the slope of the pressure ratio curves are plotted versus the specific speed (Fig. 8), the diffusion ratio (Fig. 9), the number of the blades (Fig. 10) and the blade height ratio (Fig. 11). No clear correlation between the specific speed, the diffusion ratio, the blade number and the blade height ratio and the slope of the pressure ratio is seen.

CONCLUSIONS AND DISCUSSION

In this work the effect of the tip clearance on the performance of a centrifugal compressor was studied with six different compressors. The efficiency and the pressure ratio of the compressor stages were measured, and the results were analysed together with the results found in the literature.

As mentioned in the literature, also in these measurements the pressure ratio and efficiency decreased as the tip clearance increased. The efficiency and the pressure drops in respect with the increasing tip clearance were almost linear. The slopes of these efficiency end pressure drop curves were different with different

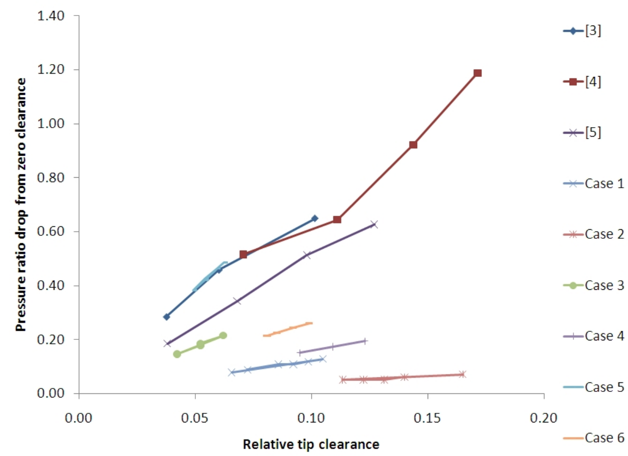

Figure 7. Pressure ratio versus relative tip clearance.

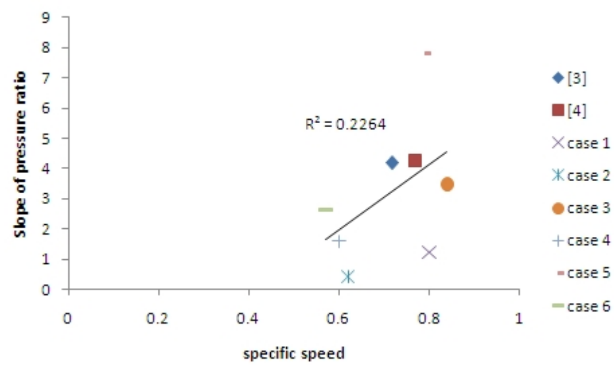

Figure 8. Slope of pressure ratio versus specific speed.

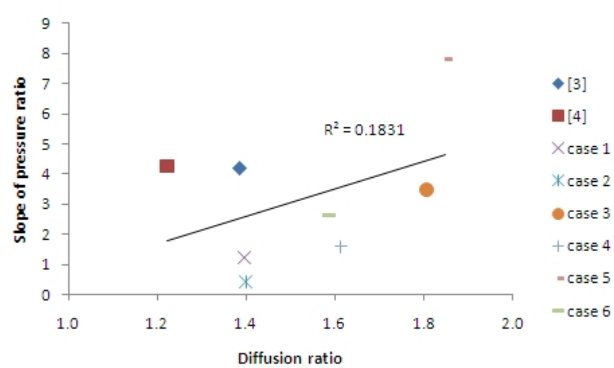

Figure 9. Slope pressure ratio versus diffusion ratio.

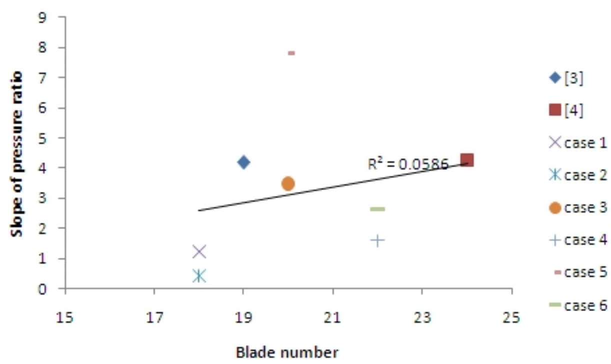

Figure 10. Slope pressure ratio versus number of blades.

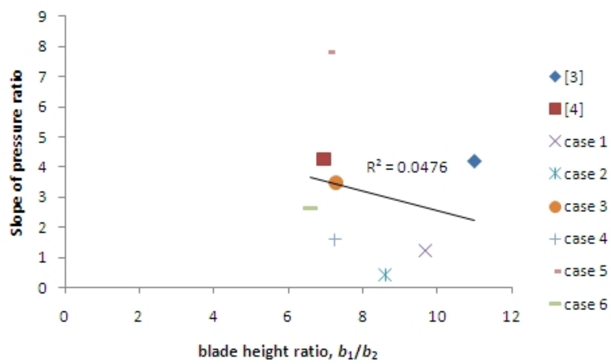

Figure 11. Slope pressure ratio versus ration of blade heights.

compressors and the drop in the efficiency was altering from 0.02 to 0.1 with the increment of 0.1 in relative tip clearance ratio.

In order to find suitable parameters to evaluate the effects of tip clearance in the compressor design, the decrease in the efficiency and the pressure ratio due to the tip clearance were studied against the diffusion ratio, the specific speed, the number of blades and the ratio of blade heights. No correlation was found for the efficiency against the specific speed. The correlation is not clear either for the diffusion ratio. The compressor with the higher diffusion ratio tends to be more sensitive for the tip clearance but the deviation in the results is large.

As for the pressure ratio, no clear correlation was found for the slope of pressure ratio with respect of the specific speed, the diffusion ratio, the blade number and the ratio of blade heights.

All in all, no clear correlation of studied parameters was found. There might be several reason for this. One is that the whole compressor stage performance was monitored and there is uncertainty about the change in the performance of other parts of the compressor than the rotor. It is inevitable that by changing the tip clearance, the flow field or the geometry of the diffuser changes. Also, the tip clearance loss is rather complex phenomena, and the parameters that describe the flow field might be more suitable than the parameters describing the overall operation of the compressor. In order to obtain such parameters, detailed flow field measurements in the compressor are needed.

It is evident that simple tip clearance models can not accurately take into account the different parameters effecting the tip clearance losses. However, in order to improve preliminary design of the impellers, rather simple models are needed. These models can be used in the meanline design code to take into account different parameters affecting to the tip clearance loss.

More research is needed to find out suitable parameters which could better predict the effect of tip clearance on the compressor performance.

ACKNOWLEDGMENT

The authors would like to acknowledge the financial contribution of the Academy of Finland.

REFERENCES

[1] Pampreen, R., 1973. "Small turbomachinery compressor and fan aerodynamics". *Journal of Engineering for Power*, July, pp. 251–256.

[2] Mashimo, T., Watanabe, I., and Ariga, I., 1979. "Effects of fluid leakage on performance of a centrifugal compressor". *Journal of Engineering for Power,* **101**, July, pp. 337–342.

[3] Klassen, H. A., Wood, J. R., and Schumann, L. F., 1977. Experimental performance of a 16.10-centimeter-tip-diameter sweptback centrifugal compressor designed for a 6:1 pressure ratio. Tech. Rep. X-3552, Nasa, june.

[4] Klassen, H. A., Wood, J. R., and Schumann, L. F., 1977. Experimental performance of a 13.65-centimeter-tip-diameter tandem-bladed sweptback centrifugal compressor designed for a pressure ratio of 6. Tech. Rep. 1091, NASA, november.

[5] Eisenlohr, G., and Chladek, H., 1994. "Thermal tip clearance control for centrifugal compressor of an apu engine". *Journal of Turbomachinery,* **116**, October, pp. 629–634.

[6] Palmer, D. L., and Waterman, W. F., 1995. "Design and development of an advanced two-stage centrifugal compressor". *Journal of Turbomachinery,* **117**(2), Apr., pp. 205–212.

[7] Eckert, B., and Schnell, E., 1961. *Axial- und Radial-Kompressoren*, 2nd ed. Springer-Verlag.

[8] Pfleiderer, C., 1961. *Die Kreiselpumpen für Flüssigkeiten und Gase*, 5th ed. Springer-Verlagen.

[9] Senoo, Y., and Ishida, M., 1987. "Deterioration of compressor performance due to tip clearance of centrifugal im-

pellers". *Journal of Turbomachinery,* **109**, January, pp. 55–61.

[10] Dean, R., and Senoo, Y., 1960. "Rotating wakes in vaneless diffuser". *Journal of Basic Engineering,* **82**.

[11] Eckardt, D., 1975. "Instantaneuos measurements in the jet-wake discharge flow of a centrifugal compressor impeller". *Journal of Engineering for Power,* **97**, pp. 337–346.

[12] Krain, H., 1981. "A study on centrifugal impeller and diffuser flow". *Journal of Engineering for Power,* **103**, pp. 688–697.

[13] Krain, H., 1988. "Swirling impeller flow". *Journal of Turbomachinery,* **110**, pp. 122–128.

[14] Schleer, M., and Abhari, R. S., 2008. "Clearance effects on the evolution of the flow in the vaneless diffuser of a centrifugal compressor at part load condition". *Journal of Turbomachinery,* **130**, p. 9.

[15] ISO 5389, 1992. *Turbocompressors - Performance test code*. International Stardardization Organization.

[16] Daily, J. W., and Nece, R. E., 1960. "Chamber dimension effects on induced flow and frictional resistance of enclosed rotating disks". *Journal of Basic Engineering*, pp. 217–232.

[17] Dibelius, G., Radtke, F., and Ziemann, M., 1984. "Experiments on friction, velocity and pressure distributions of rotating disks". Proceedings of the International Centre for Heat and Mass Transfer, pp. 117–130.